Performance of thermal distribution systems in large commercial buildings

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Abstract

This paper presents major findings of a field study on the performance of five thermal distribution systems in four large commercial buildings. The five systems studied are typical single-duct or dual-duct constant air volume (CAV) systems and variable air volume (VAV) systems, each of which serves an office building or a retail building with floor area over 2,000 m^2. The air leakage from ducts are reported in terms of effective leakage area (ELA) at 25 Pa reference pressure, the ASHRAE-defined duct leakage class, and air leakage ratios. The specific ELAs ranged from 0.7 to 12.9 cm^2 per m^2 of duct surface area, and from 0.1 to 7.7 cm^2 per square meter of floor area served. The leakage classes ranged from 34 to 757 for the five systems and systems sections tested. The air leakage ratios are estimated to be up to one-third of the fan-supplied airflow in the constant-air-volume systems. The specific ELAs and leakage classes indicate that air leakage in large commercial duct systems varies significantly from system to system, and from system section to system section even within the same thermal distribution system. The duct systems measured are much leakier than the ductwork specified as “unsealed ducts” by ASHRAE. Energy losses from supply ducts by conduction (including convection and radiation) are found to be significant, on the scale

similar to the losses induced by air leakage in the duct systems. The energy losses induced by leakage and conduction suggest that there are significant energy-savings potentials from duct-sealing and insulation practice in large commercial buildings.

1 Introduction

According to the Commercial Buildings Energy Consumption Survey (EIA 1997), there were approximately 4.6 million commercial buildings containing about 5.5 billion square meters of floor area in the United States in 1995. About 650 billion kWh of energy is used for space conditioning and ventilation in all commercial buildings annually. Commercial buildings with floor areas over 2,000 m^2, account for about 13% of all commercial buildings, corresponding to over 60% of the total floor area of the commercial buildings. In 1997, California State alone used over 8,800 GWh of electricity in operating fans and pumps for air conditioning systems (CEC 1998). Understanding the performance of thermal distribution systems and identifying perspectives of system-efficiency improvement is obviously important.

A previous study finds that leakage airflow through duct systems in light commercial buildings equals approximately one quarter of system fan-flow (Delp et al. 1998a). Underestimation of air leakage and heat conduction may lead to inappropriate HVAC system sizing and design, e.g., excessive fan-power requirement, which results in inefficient operation of HVAC equipment. Compared to light commercial buildings, a much larger fraction of HVAC energy use in large commercial buildings is associated with fan energy use, which is dramatically impacted by air leakage and conduction losses (Modera et al. 1999). Based upon computer simulation of a variable air volume (VAV) system with a leakage class of 137, Franconi et al. (1998) report an HVAC energy cost increase of 14% and an annual fan energy use increase of 55% due to the leakage alone. This suggests that sealing duct leaks in large commercial buildings would be increase energy-delivery efficiency of thermal distribution systems. Other benefits to airtight duct systems in large buildings include better control of airflow at the registers (or flow balancing), and providing potentially better indoor air quality and thermal comfort.

Compared to the research on duct systems of residential and light commercial buildings, there exists very limited study on thermal distribution systems in large commercial buildings. A field study (Fisk et al. 1998) reports that duct system leakage classes range from 60 to 270 in two large commercial buildings. These values are generally well above the ASHRAE value of 48 for “unsealed” rectangular metal ducts (ASHRAE 1997). However, the ASHRAE values, specified for different duct types instead of duct systems, neglect leakage at connections of ducts to grilles, diffusers, registers, duct-mounted equipment, or access doors. The air leakage ratios (up to 30%) of supply fan flow indicate that air leakage could induce significant thermal energy losses during the transportation of conditioned air through duct systems. In the same study, significant thermal losses due to heat conduction through duct walls were also uncovered. Due to the very limited number of systems studied, there is, however a lack of information about the performance of thermal distribution systems, especially in the sector of large commercial buildings. To further understand the performance of thermal distribution systems in large commercial buildings, it is necessary to assess the operating performance of actual systems in more large commercial buildings.

2 Objectives

The study aims at advancing the state of knowledge about the operating performance and energy losses of thermal distribution systems in large commercial buildings, and identifying opportunities of system-efficiency improvement. The results will eventually help construction and energy services industries to reduce the energy waste associated with thermal distribution systems in large commercial buildings. The specific objectives are 1) to assess air leakage through duct systems, measuring both effective leakage area (ELA) and operating pressures, and to estimate the ratio of leakage airflow rates; and 2) to assess the magnitude of conduction heat gains (during cooling) and/or heat losses (during heating) through duct systems in normal operation.

3 Approach

The main approach is to obtain field data on the thermal performance of duct systems in large California commercial buildings, including characterizations of the spaces in which those ducts are located. The performance evaluation of thermal distribution systems includes measurements of air leakage, pressures, and temperatures of the duct systems. Tracer gas method was used to measure total fan flow in the constant-air-volume (CAV) systems. Since the buildings in this study were generally occupied, the tests had to be as non-obtrusive as possible. This required working outside of the normal (daytime) schedules of the occupants. Studies on each of the systems included contacts with building managers and engineers; system characterization by walk-throughs and literature review; measurements of air leakage, pressure, airflow, and heat gain or loss; and data analyses. The following describes the measurements used in this study.

3.1 Effective leakage area

To characterize the airtightness of thermal distribution systems, the effective leakage areas (ELAs) of isolated sections of duct systems were measured using fan-pressurization procedures. The ELA is defined as the area of a perfect nozzle (i.e., orifice) that, at some reference pressure difference, would produce the same flow as that passing through all the leaks in the system. By artificially creating a series of pressure differences across the leaks, the ELA can be determined by fitting the flow and pressure data to Eq. (1):

$$Q = \frac{ELA}{10^4}\sqrt{\frac{2\,\Delta P_{\text{ref}}}{\rho}}\left(\frac{\Delta P}{\Delta P_{\text{ref}}}\right)^n , \tag{1}$$

where Q is the volumetric flow rate ($m^3\ s^{-1}$), ELA is the effective leakage area (cm^2), ΔP is the pressure difference across the leaks in the system (Pa), ΔP_{ref} is a reference pressure difference (Pa), n is the pressure exponent (-), and ρ is the air density (kg m^{-3}).

The method is well documented in the literature (SMACNA 1985; ASTM 1987, Delp et al. 1997). In order to measure the ELAs and register airflow in some of the large systems, we developed the turbo-blaster (Xu et al. 1999a), a variable-speed fan with an integral airflow meter, for the use of injecting air into the isolated section of large duct-systems for ELA measurement, or creating the “quasi-zero” pressures in the fan-powered flowhood connected to individual registers for register airflow measurement. The flow rates through variable-speed fan were recorded at various levels while the pressure differences between the interior and exterior of the duct were monitored simultaneously. The pressure differences recorded ranged between 10 and 200 Pa.

The pressure exponent typically has a value near 0.6. Given the uncertainties in measured air injection rates and average measured pressure across leaks[1], the uncertainty in the measured ELA is estimated to be about ±8% using the duct-blaster, and ±6% using the Turbo-blaster

[1] The accuracy of the flow sensor integral to the fan is ±3% for the Duct-blaster (±1% for the Turbo-blaster). We assume that the average pressure drop across leaks in the duct systems could vary by up to ±2 Pa from the

(Xu et al. 1999a). In fact, much lower ELA measurement uncertainties can be achieved in duct systems with higher operating pressures, since the reference pressure ΔP of 25 Pa is usually set in the characterization of residential and light commercial duct systems, while the operating pressures in large commercial systems exhibit much higher values.

3.2 Duct leakage class

The leakage class, C_L, is another common metric (ASHRAE 1997) used to characterize the leakage rate of per unit area of duct surface at 250 Pa pressure across the duct leaks. Using their leakage classes can compare the degrees of air leakage in duct systems of different sizes. ASHRAE (1997) lists attainable leakage classes ranging from 3 to 12 for “quality construction and sealing practices,” but notes that these attainable leakage classes do not account for leakage at connections of ducts to grilles, diffusers, registers, duct-mounted equipment, or access doors.

3.3 Duct system pressure

Operating pressures in ductwork can be significantly different from one system to another or even within one single system. Operating pressures upstream and downstream of a terminal unit (e.g. a VAV box) may vary by a factor of 10 or more. Therefore, to characterize the air leakage flows of duct systems with the ELA defined in Eq. (1), it is necessary to measure duct system pressures during normal operation. In CAV systems, static pressures across the ductwork do not vary over time given a fixed fan-speed during system operation. They were measured at multiple locations in the ductwork (e.g., plenums, branch locations, and terminal units) using handheld electronic pressure transducers with a 0.1 Pa resolution (Energy Conservatory: Pressure & Fan Flow Gauge, Model DG3, Minneapolis, Minnesota). In VAV

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average measured static pressure (using 25 Pa) in the duct during the ELA measurements, resulting in the ±8%

HVAC systems, the static pressures may likely change over time. The pressures were monitored with a data logger (Energy Conservatory: Automated Performance Testing System, Minneapolis, Minnesota) for an extended operating period (e.g., several days). These measurements covered a range of operating pressures induced by varied fan speeds and VAV damper positions.

Pressure pan measurement method has been proposed to estimate operating pressures in the ductwork in residential (ASHRAE 1999) and in light commercial building systems. In the study, we used a digital pressure gauge connected with a tube going through a sealed register-size pan, which was designed to fully block a register to obtain static pressure across the block during normal system operation. Its key advantage over the direct register pressure measurement is that it is much more repeatable, as was also indicated by Walker et al. 1998.

3.4 Airflow through registers

To measure airflow through supply registers more accurately than possible with commercially available passive flow hoods, we used an LBNL-designed, fan-powered flow hood. During the measurement, air leaving the register passes through a collection hood, then into a duct connected to a variable-speed fan equipped with an integral flow meter. The fan speed was adjusted manually to maintain a low and steady static-pressure difference between the interior of the collection hood and the room air. The flow rate was determined with the fan's integral flow meter. Different from picking a single point measurement in the study by Fisk et al 1998, we took multi-point measurements above and below the "proxy zero" pressure difference (e.g., 0±0.5 Pa) between the collection hood interior and the room. This enables us to interpolate the flow at "zero" pressure difference. We can assume that the flow rate through the register is only marginally affected by the presence of the flow hood, the

uncertainty of pressure measurements.

boundary conditions seen by the register being the same with and without the device. Note, however, that the minimum pressure drop across the register should be at least 5 Pa to limit to 5% the measurement uncertainty due to small deviations of the pressure boundary condition (Xu et al. 1999a)[(2)].

3.5 Air leakage ratio through ducts

Air leakage ratio, defined as the air leakage flow rate divided by the total airflow rate through a cross section in upstream of the ductwork, is used to characterize the degrees of air leakage from duct systems. To estimate the air leakage ratio through ducts in a CAV system, we measured the total airflow rate through a cross section in upstream of the ductwork using the tracer gas method, and measured the air leakage flow rate using two methods described as follows. For a VAV system, airflow usually changes over time. We did not perform flowrate measurement for such duct systems.

The two methods used to estimate the air leakage flow rates through duct systems are: a) to derive air leakage flow rates from ELAs and operating pressures based upon Eq. (1); and b) to estimate air leakage flow rates by taking the difference between upstream airflow rate and sum of register flow rates. To measure airflow rates through supply registers, we used LBNL-designed, fan-powered flow hoods (i.e., duct-blaster or turbo-blaster, Xu et al. 1999a) that are more accurate than using commercially available passive flow hoods.

The first method of estimating rates of air leakage is to calculate air leakage flow (Q) from Eq. (1). Ideally, it requires that the leakage areas of sections of the ductwork that operate at very different pressures be determined separately. However, the pressures monitored at limited locations may not accurately represent the actual pressure distribution in the duct

[2] This error analysis assumes that the flow through the register is proportional to the square root of the pressure difference at the boot, and that the pressure boundary condition deviates from its value of 0.5 Pa due to the

systems. This implies that the variations of the static pressures with the leak sites and/or with time are mostly unknown. The method also assumes that the discharge coefficient of the flow going through the leaks during the ELA test remains the same as that during normal operating conditions. Walker et al. (1998) have used essentially the same method to measure air leakage from residential ducts, and they estimated that the maximum uncertainty was 40% of the measured air leakage flow rate. Therefore, this approach only provides a rough estimate of the air-leakage rates.

The second method of estimating the rate of air leakage from a section of ductwork is to a) measure airflow rates through all supply registers; and (c) subtract the sum of the register flow rates from the upstream total flow rate, which is measured by using the tracer gas method. The main limitation to this approach is that the expected difference between the upstream flow rate and sum of register flow rates was comparable in magnitude to the measurement uncertainty. We might expect a ±5% uncertainty in the total register flow rate, and an ±11% uncertainty the upstream duct flow using tracer gas measurement (Xu et al. 1999a). For example, the measurement error bound in the air-leakage rate would be approximately ±15% for a duct system with an air-leakage ratio of 20%. In this case the measured air-leakage ratio would be between 5 and 35%.

3.6 Thermal losses through conduction

Thermal losses are due not only to air leakage but also to heat conduction. The assessment of conduction losses (including convection and radiation) focused on the analysis of monitored air temperatures in the system. Thermal measurements were made with stand-alone temperature loggers in the plenum (downstream of the cooling/heating coil), in selected

presence of the hood.

supply registers, in the conditioned space, in the ceiling cavity, and in the outside air. The battery-powered temperature loggers with external temperature sensors were Pro HOBOs (Onset Computer Corporation, Pocasset, MA) with 0.03 °C resolution and an accuracy of ±0.2 °C in high-resolution mode. The temperatures measured by multiple collocated temperature loggers shows a maximum span of 0.25 °C and a standard deviation of less than 0.1 °C. Studies (Delp et al. 1998a, 1998b) evaluate the energy delivery effectiveness of heat transport through ducts in terms of the duct's "cumulative effectiveness," defined as the ratio of the energy delivered at the register to the potential available at the plenum (upstream of conduction losses). Since latent heat due to moisture contents could be negligible (e.g., in supply duct latent heat is zero during operation because the duct is normally pressurized), it equals the ratio of the sensible heat capacity for heating or cooling delivered at the register to the capacity available at the plenum. Based on the assumptions that the airflow through the ductwork is constant over time and space, and impact of leakage flow on temperature change is negligible, it can be simplified by calculating the temperature differential between the register temperature, plenum temperature and the reference temperature which is essentially the conditioned-space temperature.

For VAV systems, the airflow rates usually change over the course of a day. Although the assessment on energy delivery effectiveness has to be linked to the airflow rates over a period of time (e.g., a day), "temperature effectiveness" (Xu et al. 1999b) indicates the degree of delivery effectiveness for a shorter period of time during which the airflow can be considered constant. The cumulative temperature effectiveness is the ratio of the temperature difference between the terminal units and the conditioned space to the temperature difference between the supply plenum and the conditioned space during a certain period of time. Eq. (2)

defines the *cumulative temperature effectiveness* $\tau_{s,i}(t')$ for heating or cooling, which serves as an indicator for temperature gain/loss induced by heat conduction through system ducts:

$$\tau_{s,i}(t') \equiv \frac{\int_0^{t'} \left[T_{\text{terminal unit},i}(t) - T_{\text{room}}(t)\right] dt}{\int_0^{t'} \left[T_{\text{plenum}}(t) - T_{\text{room}}(t)\right] dt}, \tag{2}$$

where t' is the elapsed period of time of interest, normally a combination of temperature swings; $T_{\text{terminal unit, i}}$ (t) is the temperatures of supply terminal unit i at time t (°C); $T_{\text{room}}(t)$ is the room temperature at time t (°C); $T_{\text{plenum}}(t)$ is the supply-plenum temperature at time t (°C). Under stable airflow conditions, cumulative temperature effectiveness is equivalent to the ratio of the sensible heat capacity (energy) for heating or cooling delivered at the supply terminal unit to the capacity available at the plenum over a cumulative period of time, which equals the “cumulative effectiveness” used in previous studies (Delp et al. 1998a). However, in general, the *temperature effectiveness* does not directly indicate energy delivery efficiency for VAV systems with or without induction units.

4 Results

We conducted field characterization testing on five HVAC systems (or system sections) in four large commercial buildings in northern California. Field study results include the physical characteristics of buildings and building systems, air leakage assessments using effective leakage areas (ELAs), air leakage classes, static pressures, and air leakage ratios, and evaluation of thermal losses due to heat conduction. The systems tested include CAV, VAV systems, and a dual-duct system with mixing boxes. Three large office buildings and one supermarket building containing these systems are characterized.

4.1 Large commercial buildings and systems

System L1, with 130 kW of cooling capacity, has constant airflow in supply and return ducts, serving the spaces of a supermarket store with a space area of 5,125 m^2. System L2 is a perimeter heating supply duct with a heating capacity of 12.8 kW. The heating duct is 60 m long, and serves one floor of perimeter offices in a large building with a total area of 2,183 m^2. This building has four such heating systems and is connected to another building of the same use and of similar floor plan. System L3 is a variable air volume system with a maximum cooling capacity of 141 kW provided by two compressors. The system has induction units at some of the VAV branches serving the core office spaces of the same building as System L2 does. System L4 is also a VAV system with a maximum cooling capacity of 484 kW. It has induction units at different VAV boxes serving spaces of an office building with a floor area of 6,075 m^2. The VAV systems (systems L3 and L4) have few return ducts in their ceiling plenums. System L5 (with the maximum cooling capacity of 352 kW) is a dual-duct system with mixing boxes downstream of the heating and cooling ducts to serve the office spaces of 3,198 m^2 . Floor area per supply register ranged from 5.2 to 29.3 m^2 for office buildings, and was 176 m^2 for the supermarket store, which housed many freezers with significant internal cooling.

4.2 Effective leakage area, air leakage class, static pressure, and air leakage ratio

ELAs and static pressures were measured for five systems or their sections in four large commercial buildings. The system sections were selected for the VAV systems or dual-duct systems on the basis of physical accessibility.

For System L1, the supply and return ducts were tested separately. System L2 is a single-duct perimeter system serving an office building, of which we tested ELAs for the supply duct, and the whole duct system. System L3 contains section L3a, the main duct upstream of the VAV boxes and induction units in the office building, and section L3b, one of the branches downstream of a VAV box with an induction unit. Section L4a and L4b in System L4 are two branches downstream of their VAV boxes with induction units in an office building. Sections a-d of System L5 are four branches downstream of their mixing boxes in an dual-duct system of another office building.

The measured effective leakage areas, air leakage classes, and static pressures in large commercial building systems are summarized in Table 1.

4.2.1 Supply duct ELA at 25 Pa (ELA_{25})

The specific effective leakage area (ELA_{25}), defined as the measured ELA at 25 Pa divided by the duct surface area and the served floor area, respectively, is used to compare the degree of air leakage for duct systems (or sections) of different sizes. The specific ELA_{25} of supply ducts varied widely from system to system, ranging from 0.7 to 12.9 cm^2 per m^2 of duct surface area, and from 0.1 to 7.7 cm^2 per square meter of floor area served. In System L3, the specific ELA_{25} per duct surface area of the section upstream of the VAV boxes was found to be eight times smaller than that of the downstream branches. Specific ELAs of the four sections downstream of mixing boxes in system L5 were much larger than those of the other systems tested: their ELAs ranged from 7.8 to 12.9 cm^2 per m^2 of duct surface area, and from 2.0 to 7.7 cm^2 per m^2 of floor area served. Overall, the findings from the systems indicate a much wider range of specific ELAs than those reported by Fisk et al. 1998, which ranged between 1.0 and 4.8 cm^2 per m^2 of duct surface area.

4.2.2 Air leakage class

As another way to assess air leakage of duct systems, ASHRAE-defined air leakage class is calculated for the duct systems and sections tested. Air leakage classes for the main supply ducts (upstream of VAV or mixing boxes) for all systems tested ranged from 34 to 246, while those downstream (usually branches) varied much more widely, from 58 to 606. Not unexpectedly, the data showed significant differences of air leakage classes between duct systems, and between sections downstream of VAV boxes, or mixing boxes. Overall, the leakage classes of all duct sections (including return ducts) ranged from 34 to 757. The median leakage class of all samples presented is about 300. These values are much higher than the leakage classes predicted by ASHRAE for unsealed ducts, which ranged from to 30 to 48.

4.2.3 Operating pressure

Usually VAV systems had higher operating pressure in the main ducts than the CAV systems. The average supply-plenum pressures of the two VAV systems (L3, L4) ranged between 480 and 610 Pa, while the supply-plenum pressure of a main CAV system (L1) was 245 Pa. Another dual-duct CAV system (L5) had the hot deck pressure of up to 145 Pa and the cold deck pressure of up to 80 Pa when in operation. The perimeter CAV system, L2, had a plenum operating pressure of 79 Pa.

The static pressures of ducts downstream of terminal boxes (VAV or mixing boxes) ranged from 16 to 47 Pa during normal operation. Duct sections or branches downstream of terminal boxes had an average operating pressure of approximately 35 to 39 Pa. Therefore, the average operating pressures varies significantly among different systems (e.g. types), and even among different sections of the same systems.

4.2.4 Air leakage ratio

We examined airflow leakage rates in two of the CAV systems (L1 and L2) using two different approaches: 1) derivation by the measured ELA and operating pressure, and 2) flow-subtraction method. Similar to the pressure pan measurement drafted in the proposed ASHRAE Standard 152P, taking the average of pressure pan measurements for all supply registers is an estimate of the average operating pressure in the supply duct of a large CAV system. On the other hand, taking half of the value measured in the large-duct supply-plenum is another way to estimate the operation pressure, which was also adopted for characterizing duct systems in residential and light commercial buildings. The total fan flow rates in sectional supply ducts (e.g., main ducts) are measured by the tracer gas technique. Air leakage ratio is then obtained by dividing the leakage flow rate by the total fan flow rate. Table 2 provides rough estimated air leakage ratios for Systems L1 and L2. Using the first approach (ELA and operating pressure), the estimate of the supply section's air leakage ratio is 10% based on the pressure-pan measurements, and is 21% if using the half plenum pressure as the input for operating pressure. The estimated air leakage ratio for the return section is 6% if based on the pressure-pan measurements, and is 23% based on the method of half plenum pressure. As discussed in the approach section, Walker et al. (1998) have used essentially the same method to estimate air leakage from residential ducts, and they estimated that the maximum uncertainty was 40% of the measured air-leakage rate.

By using the flow-subtraction approach, the estimation of the leakage ratio for supply section in system L1 is 3%, which is associated with the combined uncertainty of ±16%. For system L2, the estimated air leakage ratio for the supply section is 26% based on the method of half plenum pressure, and based on the average pan pressure method. By using the flow-subtraction method, we measured a leakage ratio of 17% associated with the uncertainty of ±16%.

With the uncertainties pertaining to the estimation, comparisons between the two approaches indicate that the leakage ratio would be in the range from zero to 19% for the supply duct of system L1, and in the range from zero to 33% for the supply duct of system L2.

Overall, given the uncertainties associated with the two different methods used in this study, the range of the estimated leakage ratios in System L1 and L2 is between zero to a third of the total fan flow. This is similar to the findings by Fisk et al. 1998, which report that the estimated air-leakage ratios in the two large systems ranged from zero to approximately 30%.

4.3 Conduction losses through ducts.

We monitored duct air temperatures in three systems in large commercial buildings (L1, L2, and L3). Measurements were made over several days at an interval of 10 seconds to detect temperature swings. The following are the main findings in temperature monitoring and heat conduction analysis.

Figure 1 shows the temperature trend within a CAV system (L1). The registers and duct layout is shown in Figure 2. Figure 3 shows the temperature trend for System L2 with a heating-supply fan with a constant speed for perimeter offices. The registers and duct layout is shown in Figure 4. Figure 5 shows the temperature trends for System L3, with three VAV boxes and two downstream registers, which are shown in Figure 6. The temperature differences between the supply plenum and VAV boxes (or registers) indicate temperature rises throughout ductwork during cooling operation.

For some systems tested, the supply temperature swung significantly, so did the air temperature exiting the supply registers. The temperature difference between supply-registers and supply-plenum thus varied accordingly. We calculate the temperature difference between the supply plenum and terminal units (i.e., registers and/or VAV boxes) at the end of each temperature swing as a way to assess magnitudes of thermal loss through conduction in different systems. Table 3 presents average temperature rise (+, in cooling mode) or drop (-, in heating mode) relative to the supply plenum for each of the registers. The overall average values in the right column can be used as indications of the heat conduction impacts through ductwork.

For two large CAV systems (L1 and L2) tested in heating mode, the average temperature drop (at the end of each temperature swing) between the supply plenum and the supply registers ranged from 2 to 3.6 °C, while the temperature drop in individual registers ranged from 0.3 to 6.2 °C. The corresponding cumulative temperature effectiveness of downstream registers was 0.77 and 0.98 (Table 3). Within each of the systems, the further the distance downstream of the supply-plenum, the lower the cumulative effectiveness. Since the flow did not changed overtime in L1 and L2, the cumulative temperature effectiveness is equivalent to the “delivery effectiveness,” which indicates the energy delivery effectiveness of the duct system (Delp 1998a).

For the VAV system (L3) tested in cooling operation with two compressors, the supply temperatures swung periodically, as did the air temperatures exiting the supply registers. The temperature difference between the supply plenum and the terminal units (i.e., registers, VAV boxes) at the end of each temperature swing was used as a way to indicate magnitudes of thermal loss through conduction of different branches. On average, the temperature rises (at the end of each temperature swing) between the supply plenum and VAV boxes (A, B and D) ranged from 1.8 to 6.5 °C, while average temperature rises between the supply plenum and the supply registers ranged from 4.5 °C (without induction unit, Register B) to almost 12 °C (with induction unit, Register A and C). Our monitoring results of velocity pressure in the main trunk and branches suggested that during some short periods of time, the total fan flow was fairly constant. For example, between 1 PM and 2 PM, the dynamic pressure ranged from 70 to 77 Pa with an average of 74 Pa. The corresponding fan flow ranged within ±3% of its average, indicating little change in the fan flow between 1 PM and 2 PM. Since the dynamic pressure in VAV box B was quite stable during the same hour, the short-term aggregated temperature effectiveness can be used to estimate the thermal conduction loss for the specific VAV branch. We calculate the temperature effectiveness for three VAV boxes and registers to assess the magnitude of heat conduction loss for some branches and registers for the peak-hour between 1 PM and 2 PM.

Figure 7 shows the instant temperature effectiveness for one VAV box (B), and cumulative temperature effectiveness for three VAV boxes (A, B, D), and three registers (A, B, C) from around 1 PM to 2 PM. The instant temperature effectiveness of VAV box B changed periodically with temperature swings, while the cumulative temperature effectiveness achieved a relatively stable value (0.90) shortly after only one temperature-swing, which usually lasted for less than 15 minutes. For the one-hour period, the short-term cumulative temperature effectiveness was 0.90 for VAV box B, 0.73 for box A, and 0.62 for box D. This indicates that the further the VAV box was, the lower the temperature effectiveness was. The temperature effectiveness was 0.76 for register B.

Additional temperature rises in registers downstream of a VAV box rendered the temperature effectiveness of downstream registers significantly lower than that of their parent-VAV box during the peak-hour. For example, the temperature effectiveness of register B was 0.76, about additional 14-percent points' reduction for the temperature effectiveness in VAV box B (0.90). Assuming register B was representative of the registers in this particular VAV branch, the actual thermal losses (heat gains) through duct conduction downstream of VAV box B account for the additional 14-percent points of the heat gain in the branch during the peak-hour period. Assuming that the 14-percent points of the potential cooling lost is representative of the VAV branch tested, and that the average (0.82) of temperature effectiveness of VAV Box A and VAV Box B was representative of all VAV boxes in the system, there would be a total of about 32% cooling lost from supply duct due to conduction during the peak-hour.

Overall, the temperature effectiveness was between 0.77 and 0.98 for the two CAV systems tested in heating mode, with average temperature-drop of up to about 4°C. As expected, the effectiveness decreased with the distance downstream of the supply plenum. For the one VAV system in a large building in cooling mode, the temperature rise ranged between 2 °C and 6.5 °C for registers without "induction" units. Although part of the cooling losses may indirectly be transferred into conditioned zones, the losses would significantly increase the fan power consumption for supplying air.

4.4 Energy impact of leakage and conduction losses

In CAV and VAV systems, energy lost inform of air leakage or conduction to a ceiling return plenum from the supply ducts increases the amount of air that must be pushed through the fan towards the conditioned spaces to meet their loads, and thus increases fan energy consumption. This short-circuiting not only directly impacts fan-power and run-time, but also increases the cooling load whenever occurs, which is induced by the extra heat generated by the fan.

Given the magnitude of air leakage ratio and conduction losses found in this study, it would require excessive fan power to over come the leakage loss and conduction losses. If we make the following assumptions: 1) that there are 30% of thermal losses from supply air due to air leakage and conduction in the duct system; 2) that convective transport through the return plenum dominates conduction through the drop ceiling to the conditioned zone; 3) that there is no penalty associated with simultaneous heating and cooling losses to the return plenum; and 4) that the pressure drops through the distribution system and the resulting pressure differential seen by the fan are proportional to the square of the flow rate through the ducts, then 30% thermal loss would in theory result in as much as 3.4 times of original fan energy required to overcome the loss of cooling supply. Therefore, by eliminating the 30% thermal losses would have reduced the fan power required by about 70%. Despite of great potentials, dampers in VAV system would have to close down during operation, thus changing the system operating pressures. This would cause fan energy reduction due to lower flow, but not following the "cube-law." To realize the cube-law savings, we need to change the setpoint of the static operating-pressure regulator at the fan, which is limited by the minimum operating pressure requirement of the VAV boxes. For CAV systems, it is clear that an existing CAV system would not automatically adjust its airflow to accommodate a change in required airflow to the zone. Rather it would usually reset the supply air temperature, and thus no fan-power savings would be realized simply by retrofitting the ducts for CAV systems. For such CAV systems, the fan-power and energy savings could be realized by better design in new construction and systems including changing the fan speeds, or by replacing the fans in existing buildings.

5 Conclusions

The field study confirms that significant duct air leakage in large commercial buildings is common, as to what has been found in residences and light commercial buildings. Although we cannot draw any conclusions about the population of buildings based upon the five systems that we tested, it is clear that there can be significant leakage, and that there are large variations in leakage levels between and within buildings. Based upon these findings, and upon our earlier analysis of the energy implications of the leakage, the system energy losses induced by air leakage can be significant.

Thermal losses induced by conduction (including convection and radiative heat transfer) through duct-system in large buildings are also significant. The supply-temperature changes along duct-systems due to these losses ranging between 0.3 °C and 6.5 °C for branches without “induction” units. These exceeded the common assumption of 0.6°C by HVAC designers. As thermal losses induced by conduction can be similar to the losses from leakage in large commercial duct-systems, the energy-savings potential (e.g., fan, duct system) associated with these losses is significant.

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Table 1. Measured air duct system effective leakage areas, air leakage classes, and static pressures in large commercial buildings

	System L1		System L2		System L3a	System L3b	System L4a	System L4b	System L5a	System L5b	System L5c	System L5d
Duct system description	CAV		CAV		VAV	VAV	VAV	VAV	Dual	Dual	Dual	Dual
	Supply	Return	Overall	Supply duct only	Main Trunk	Branch	Branch	Branch	Duct	Duct	Duct	Duct
Year built	1996		1979		1979	1979	1980	1980	1990	1990	1990	1990
ELA per unit served floor area (cm2/m2 at 25 Pa)	0.3	0.3	0.3	0.1	-	0.7	0.3	0.3	5.1	2.0	7.7	5.0
ELA per unit duct surface area (cm2/m2 at 25 Pa)	2.5	8.8	1.9	0.7	0.7	5.4	0.9	1.3	9.9	12.9	11.5	9.7
Pressure exponent (-)	0.59	0.52	0.59	0.60	0.61	0.70	0.69	0.63	0.55	0.57	0.60	0.60
US air leakage class (cfm/100 ft^2, 250Pa†)	121	370	96	36	34	341	58	70	441	606	394	490
Plenum or terminal box pressure (Pa) *	245	-260	79	79	480	29.5*	47*	47*	50*	18*	-	16*

† Air leakage class is based on the measured duct ELA at 25 Pa and the calculated leakage flow at 250 Pa static pressure, using the measured pressure exponent.
* Average value of pressure pan measurements on all registers.

Table 2. Estimates of air leakage ratios in two large commercial building systems.

Method	Leakage ratios based on the two methods (ELA/operating pressure; flow difference between up/downstream)		
	ELA and half plenum pressure	**ELA and average pan pressure**	**Fan flow – Sum of register flows**
System L1 (supply)	21%	10%	3%
System L1 (return)	23%	6%	-
System L2	26%	26%	17%

Table 3. Temperature rise/drop and effectiveness in registers or terminal boxes.

System type		Operating Mode	Temperature rise/drop at end of heat/cooling-ON swings (°C) (Cumulative temperature effectiveness)				
			Supply register A	**Supply register B**	**Supply register C**	**Supply register D**	**Average**
Large systems	L1 CAV store	Heating	-1.5 *(0.96)*	-2.5 *(0.95)*	-	-	-2.0 *(0.96)*
	L2 CAV office	Heating	-0.3 *(0.98)*	-4.2 *(0.84)*	-6.2 *(0.77)*	-	-3.6 *(0.87)*
	L3 VAV office	Cooling	4.4 *(0.73*)*	1.8 *(0.90*)*	-	6.5 *(0.62*)*	4.3 *(0.75*)*
			11.8 *(-)*	4.5 *(0.76*)*	11.0 *(-)*	-	9.1 *(-)*

Notes:

- Data in Italics indicate the cumulative temperature effectiveness for CAV systems during normal operating hours, or for the VAV system during one peak-hour (numbers marked with "*" in parenthesis). The effectiveness can be an estimate of energy delivery effectiveness for the respective terminal units.
- Data in the shaded cells are for VAV boxes not registers.

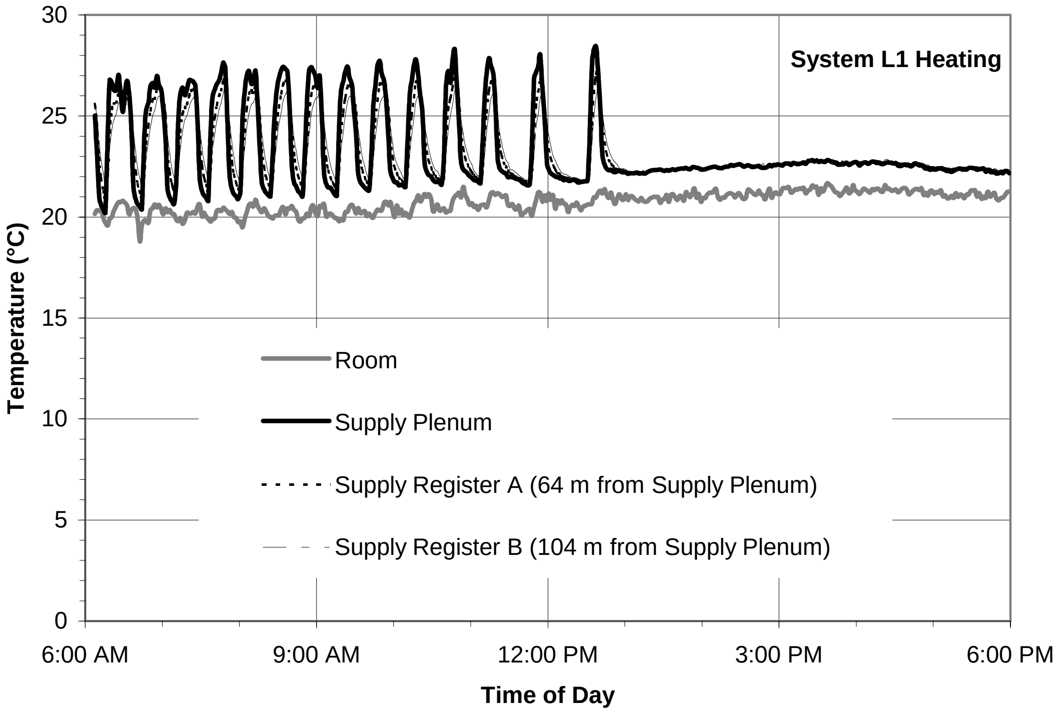

Figure 1. Temperature trend in System L1 of a supermarket store.

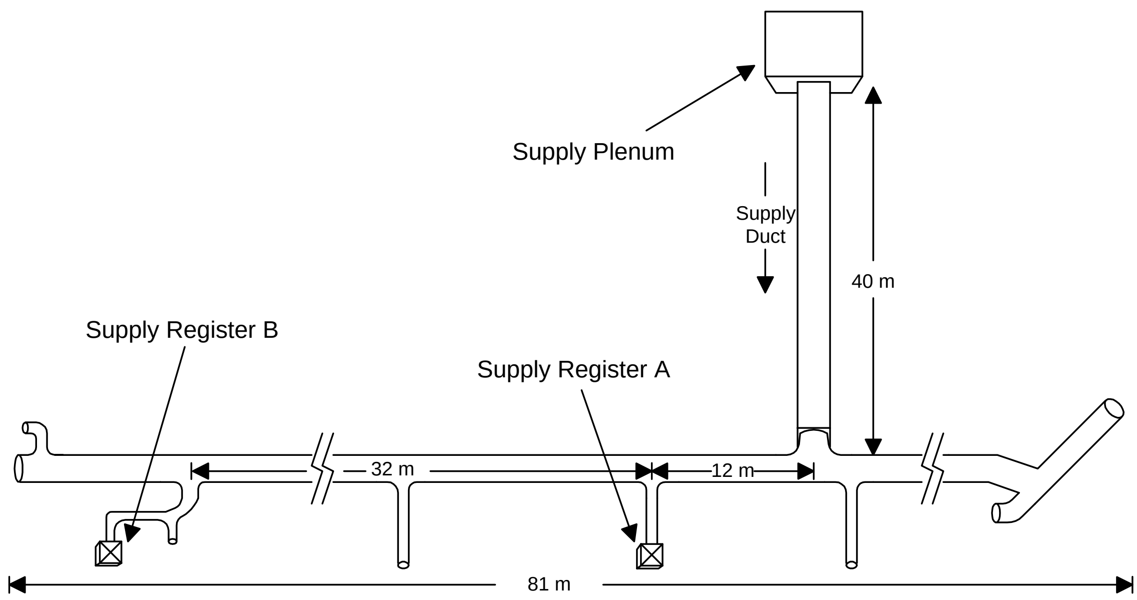

Figure 2. Registers and supply duct layout in System L1 of a supermarket store.

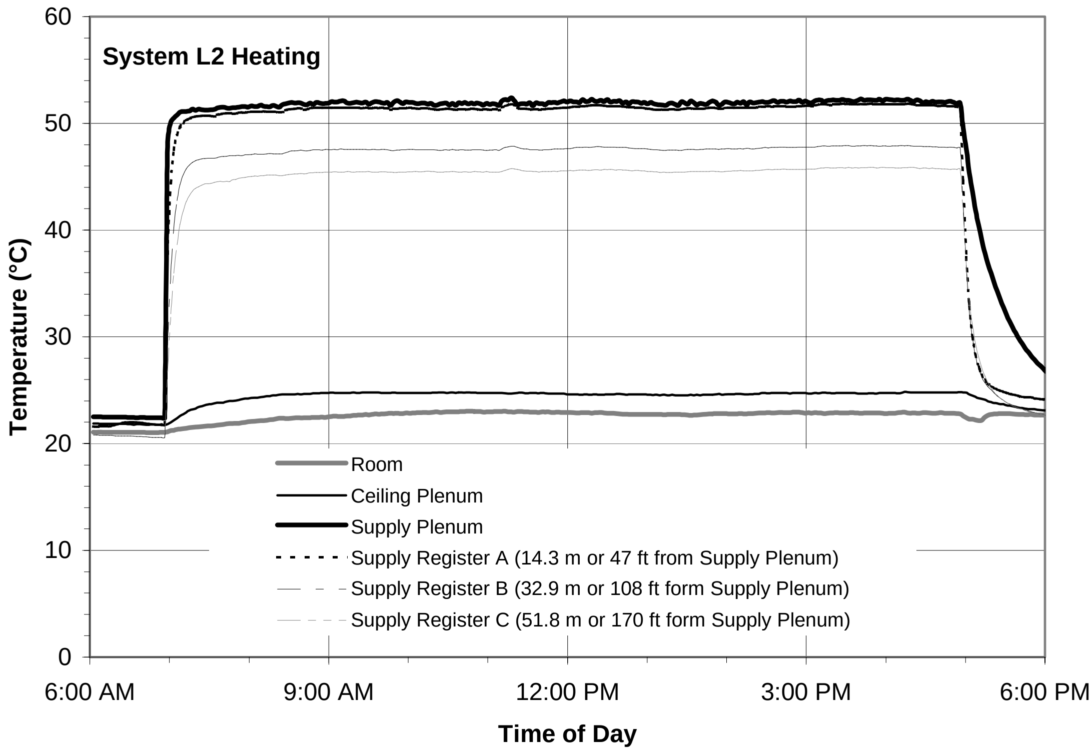

Figure 3 Temperature trend in System L2 (heating, long duct).

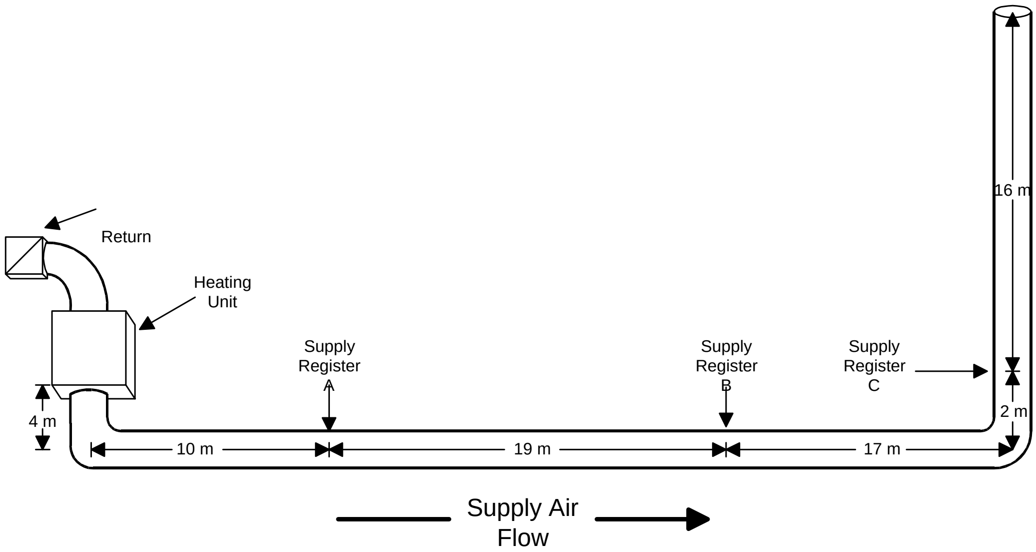

Figure 4. Registers and supply duct layout in duct System L2 (heating, long duct).

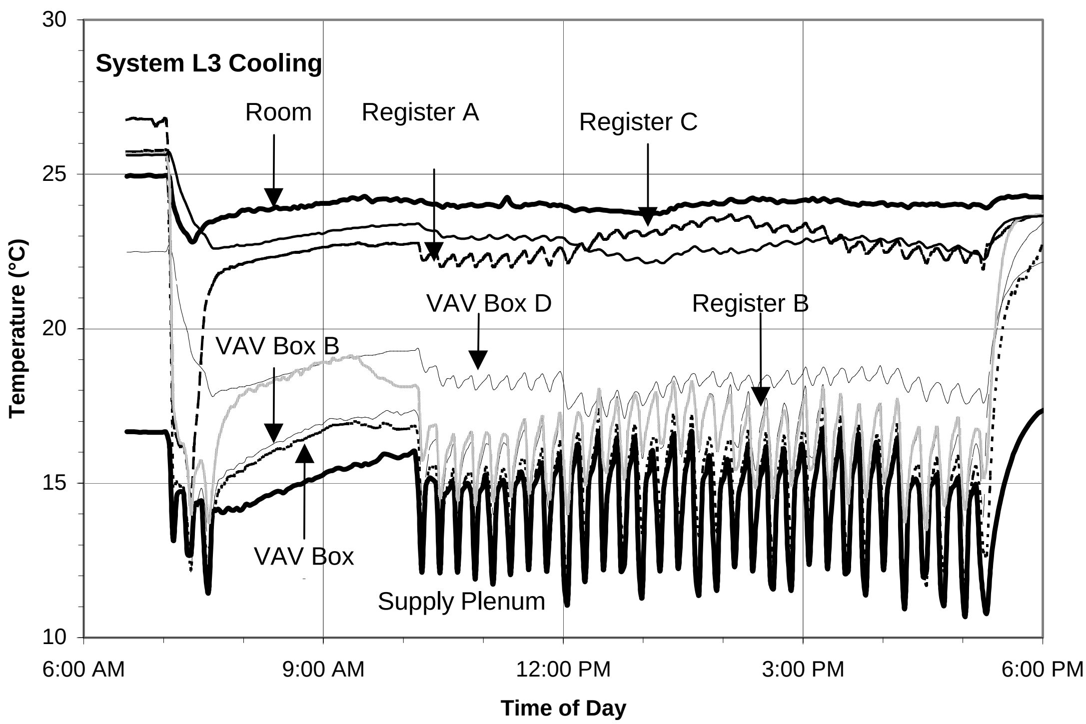

Figure 5. Temperature trends for System L3 (cooling, VAV duct system).

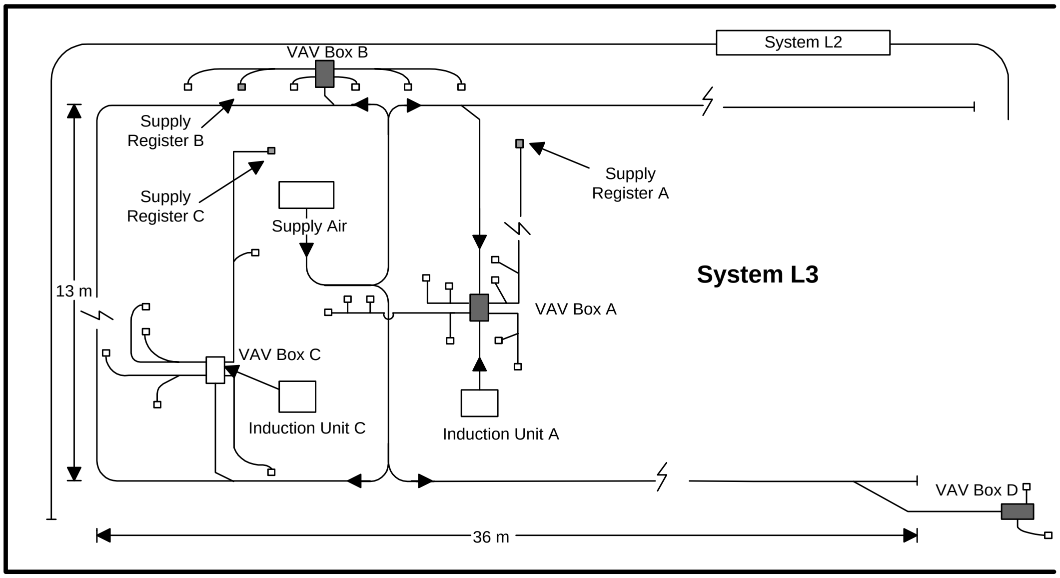

Figure 6. Monitored VAV boxes and registers layout for duct System L3.

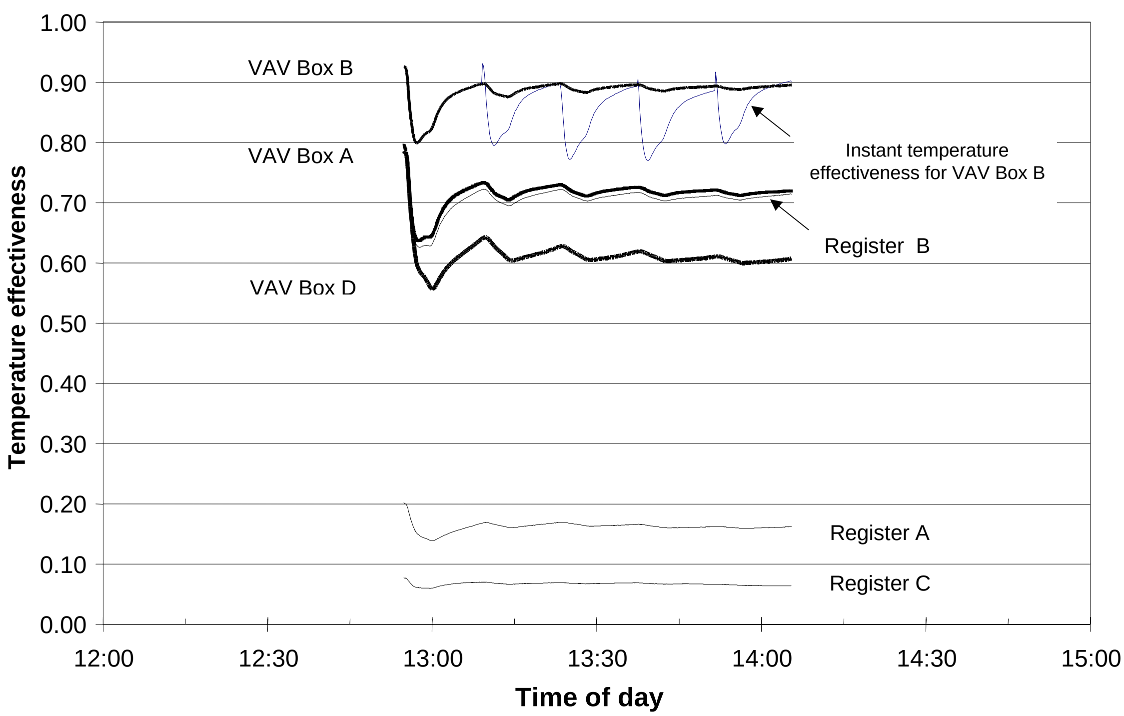

Figure 7. Short-term temperature effectiveness for VAV boxes and registers.